PATENT COOPERATION TREATY

PCT

REC'D 14 JAN 2005

WIPO PCT

INTERNATIONAL PRELIMINARY EXAMINATION REPORT

(PCT Article 36 and Rule 70)

Applicant's or agent's file reference 16PCT01	**FOR FURTHER ACTION** See Notification of Transmittal of International Preliminary Examination Report (Form PCT/IPEA/416)	
International application No. PCT/TR 03/00071	International filing date *(day/month/year)* 09.09.2003	Priority date *(day/month/year)* 09.09.2002
International Patent Classification (IPC) or both national classification and IPC F01C11/00		
Applicant AKMANDOR, Ibrahim Sinan		

1. This international preliminary examination report has been prepared by this International Preliminary Examining Authority and is transmitted to the applicant according to Article 36.

2. This REPORT consists of a total of 8 sheets, including this cover sheet.

   ☒ This report is also accompanied by ANNEXES, i.e. sheets of the description, claims and/or drawings which have been amended and are the basis for this report and/or sheets containing rectifications made before this Authority (see Rule 70.16 and Section 607 of the Administrative Instructions under the PCT).

   These annexes consist of a total of 32 sheets.

3. This report contains indications relating to the following items:

I	☒	Basis of the opinion
II	☐	Priority
III	☐	Non-establishment of opinion with regard to novelty, inventive step and industrial applicability
IV	☒	Lack of unity of invention
V	☒	Reasoned statement under Rule 66.2(a)(ii) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement
VI	☐	Certain documents cited
VII	☐	Certain defects in the international application
VIII	☐	Certain observations on the international application

Date of submission of the demand 02.04.2004	Date of completion of this report 17.01.2005
Name and mailing address of the international preliminary examining authority: European Patent Office - P.B. 5818 Patentlaan 2 NL-2280 HV Rijswijk - Pays Bas Tel. +31 70 340 - 2040 Tx: 31 651 epo nl Fax: +31 70 340 - 3016	Authorized Officer Lequeux, F Telephone No. +31 70 340-4664

Europäisches Patentamt · European Patent Office · Office européen des brevets

INTERNATIONAL PRELIMINARY EXAMINATION REPORT International application No. PCT/TR 03/00071

I. Basis of the report

1. With regard to the **elements** of the international application *(Replacement sheets which have been furnished to the receiving Office in response to an invitation under Article 14 are referred to in this report as "originally filed" and are not annexed to this report since they do not contain amendments (Rules 70.16 and 70.17))*:

Description, Pages

1-15 received on 24.12.2004 with letter of 23.12.2004

Claims, Numbers

1-6 received on 24.12.2004 with letter of 23.12.2004

Drawings, Sheets

1/11-11/11 received on 24.12.2004 with letter of 23.12.2004

2. With regard to the **language**, all the elements marked above were available or furnished to this Authority in the language in which the international application was filed, unless otherwise indicated under this item.

   These elements were available or furnished to this Authority in the following language: , which is:

   - ☐ the language of a translation furnished for the purposes of the international search (under Rule 23.1(b)).
   - ☐ the language of publication of the international application (under Rule 48.3(b)).
   - ☐ the language of a translation furnished for the purposes of international preliminary examination (under Rule 55.2 and/or 55.3).

3. With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application, the international preliminary examination was carried out on the basis of the sequence listing:

   - ☐ contained in the international application in written form.
   - ☐ filed together with the international application in computer readable form.
   - ☐ furnished subsequently to this Authority in written form.
   - ☐ furnished subsequently to this Authority in computer readable form.
   - ☐ The statement that the subsequently furnished written sequence listing does not go beyond the disclosure in the international application as filed has been furnished.
   - ☐ The statement that the information recorded in computer readable form is identical to the written sequence listing has been furnished.

4. The amendments have resulted in the cancellation of:

   - ☐ the description, pages:
   - ☐ the claims, Nos.:
   - ☐ the drawings, sheets:

5. ☒ This report has been established as if (some of) the amendments had not been made, since they have been considered to go beyond the disclosure as filed (Rule 70.2(c)).

   (Any replacement sheet containing such amendments must be referred to under item 1 and annexed to this report.)

   see separate sheet

6. Additional observations, if necessary:

IV. Lack of unity of invention

1. In response to the invitation to restrict or pay additional fees, the applicant has:

   ☒ restricted the claims.

   ☒ paid additional fees.

   ☐ paid additional fees under protest.

   ☐ neither restricted nor paid additional fees.

2. ☐ This Authority found that the requirement of unity of invention is not complied with and chose, according to Rule 68.1, not to invite the applicant to restrict or pay additional fees.

3. This Authority considers that the requirement of unity of invention in accordance with Rules 13.1, 13.2 and 13.3 is

   ☐ complied with.

   ☒ not complied with for the following reasons:

   see separate sheet

4. Consequently, the following parts of the international application were the subject of international preliminary examination in establishing this report:

   ☒ all parts.

   ☐ the parts relating to claims Nos. .

V. Reasoned statement under Article 35(2) with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1. Statement

Novelty (N)	Yes:	Claims	1-3,5-7
	No:	Claims	4
Inventive step (IS)	Yes:	Claims	1-3,5-7
	No:	Claims	4
Industrial applicability (IA)	Yes:	Claims	1-7
	No:	Claims	

2. Citations and explanations

   see separate sheet

Re Item I
Basis of the report

The figures 5,6 and 11 (and the related references in the claims 1-3 and description) have been considered as introducing amendments going beyond the disclosure as filed (Rule 70.2(c)) and have therefore not been taken into account during the examination.

Re Item IV
Lack of unity of invention

1. This Authority considers that there are **2** inventions covered by the claims indicated as follows:

    a. Claims 1-4 directed to a method of operating a sliding vane rotary engine,
    b. Claims 5-7 directed to a compound engine comprising a sliding vane rotary machine.

2. The reasons for which the inventions are not so linked as to form a single general inventive concept, as required by Rule 13.1 PCT, are as follows:

    the prior art has been identified as document EP-A-1016785 and discloses (fig 13 and column 15, lines 22 - column 16, line 12) a combustion engine comprising a vane compressor, a vane motor and an external combustion chamber wherein the inner radial surfaces of the compressor and motor chambers have a cycloidal (column 6, lines 35-43) shape.

    It follows that the special technical features "STF" (Rule 13.2 PCT) of the set of claims 1-4 making a contribution as a whole over the prior art are the specific characteristics of the method of operating, of controlling and of monitoring of the engine and its associated thermodynamic cycle.

    The STF of the set of claims 5-7 are the specific characteristics of the assembly of sliding vane machines and the associated driving and driven axial flow machines.

    In conclusion, the groups of claims are not linked by common or corresponding special technical features neither by corresponding effects and define 2 different inventions not linked by a single general inventive concept. Hence the application does not meet the requirements of unity of invention as defined in Rules 13.1 and 13.2 PCT.

Re Item V

Reasoned statement with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

Reference is made to the following documents:
D1: EP1016785
D2: US5692372

1. Claims 1-3

The document D1 is regarded as being the closest prior art to the subject-matter of claims 1-3, and shows (the references in parentheses applying to this document) an external combustion engine (fig 13 and column 15, lines 22 - column 16, line 12) comprising a vane compressor, a vane motor and an external combustion chamber wherein the inner radial surfaces of the compressor and motor chambers have a cycloidal (column 6, lines 35-43) shape and operating accordingly to a thermodynamic cycle.

The subject-matter of claims 1-3 differs from this known method of operating in that the claimed method further includes controlling and monitoring and also includes a power expansion phase up to ambient pressure and a limited temperature constant volume combustion followed by a constant pressure combustion and/or a constant temperature combustion.

The subject-matter of claims 1-3 is therefore new (Article 33(2) PCT).

Furthermore and although all individual thermodynamic steps or portions of a step of the method of claims 1-3 are already disclosed in various prior art documents (see international search report and documents cited in the application), the subject-matter of claims 1-3 is inventive (Article 33(3) PCT) as it wouldn't be obvious to the skilled person to combine, in the corresponding sequence, all the features derived from the various documents with those of D1 and to apply them to a method of operating, controlling and monitoring a sliding vane rotary engine with the corresponding technical features.

The subject-matter of claims 1-3 is industrially applicable (Article 33(4) PCT).

2. Claims 4

As the subject-matter of claim 4 is directed to an "apparatus" (namely a sliding vane engine), the "method" related features disclosed in the characterizing portion of claim 4 "characterised ... claim 3" do not contain any clear and unambiguous technical features directed to an apparatus and thus, the subject-matter of claim 4 is limited to the features of the preamble of claim 4.

As D1 discloses (see point 1 here above) a sliding vane rotary heat engine with all the features of the preamble of claim 4, the subject-matter of claim 4 is not new (Art 33(2) PCT).

The subject-matter of claim 4 is industrially applicable (Art 33(4) PCT).

3. Claims 5-7

The document D2 is regarded as being the closest prior art to the subject-matter of claim 5, and shows (the references in parentheses applying to this document) a compound propulsion engine comprising a first spool composed of a first axial flow compressor (fig 1, ref 13) drivingly interconnected via a first shaft (36) with an axial flow turbine (34) and also comprising a second spool composed of a second axial flow compressor (23) drivingly interconnected via a second shaft (36) with a rotary internal combustion Wankel engine (26) and wherein the first and second shafts are concentric (column 3, lines 32-34).

The subject-matter of claim 5 differs from this known engine in that the second compressor is of the sliding vane type, in that the rotary engine is a sliding vane type turbine, in that the first and second shafts are not concentric and in that the first and second spools are only aero-thermodynamically coupled.

The subject-matter of claim 5 is therefore new (Article 33(2) PCT). Furthermore, as the combination of features of claim 5 as such is not rendered obvious by the available prior art, the subject-matter of claim 5 is inventive (Art 33(3) PCT).

The subject-matter of claim 5 is industrially applicable (Art 33(4) PCT).

Claims 6 and 7 are dependent on claim 5 and as such also meet the requirements of the PCT with respect to novelty, inventive step and industrial applicability.

TURBO-ROTARY, COMPOUND AND SOLO PROPULSION ENGINE AND THERMODYNAMIC CYCLE

FIELD OF THE INVENTION

This invention relates to energy systems and more particularly to componentry and thermodynamic cycle for enabling shaft work, propulsion drive, electric power source, jet propulsion and thermodynamic systems such as ventilation, cooling, heat, pressure or vacuum generating systems.

BACKGROUND OF THE INVENTION

Since the start of the industrial revolution, the reciprocating piston engine based on the Otto and Diesel cycles and, the gas turbine engine based on the Brayton cycle, have largely dominated the market. Despite this fact, for many years, patents on rotary vane combustion engines have claimed that rotary engines possess many advantages over reciprocating engines such as having high torque, fewer parts, lower weight and fewer reciprocating imbalance. Fundamental design characteristics of the present invention addresses the main problems related to rotary vane engines and bridges the mass flow and rotational speed gaps between reciprocating and gas turbine engines.

Rotary Engines with Sliding Vane Slicing Through Rotor

Instead of having hinged vanes (USPTO 5,352,295, Chou Yi, October 4th, 1994 and PCT WO 02/31318, VADING Kjell, April 18th, 2002), some of the prior arts do use sliding vanes slicing through the rotor (USPTO 4,414,938, UMEDA Soei, November 15th, 1983 and USPTO 4,422,419, UMEDA Soei, December 27th, 1983; USPTO 5,596,963, LAI, Jui, H., January 28th 1997). In these arts, a plurality of spring-loaded vanes are used against the housing wall to achieve air-tightness. Furthermore, above-mentioned prior arts do not have any central vane retention mechanism (138,139,150) that would prevent the related wear problem. Moreover, only a portion of the entire inner peripheral of the housing is elliptic. Another patent, related to rotary heat engine with 'all-through solid' vane (USPTO 5,511,525, JIRNOV and al.,

April 30th, 1996), uses at least two mutually perpendicular vanes with radially extending guide. The plural use of vanes within one compressor housing substantially reduces the pressure ratio This leads to a reduction of the rotary component efficiency and also increase the system complexity as more stage is required. Furthermore, the vane guide path mechanism described in this prior art is an additional cause for increased friction wear.

Rotary Engines with Separate Compression and Expansion Chambers

There are many rotary engine patents which provide separate compression, combustion and expansion chambers (PCT WO 02/31318, VADING Kjell, April 18th, 2002; PCT WO 99/041141, O'BRIEN Kevin, January 28th 1999; USPTO 5,596,963, LAI, Jui, H., January 28th 1997; USPTO 5,335,497, MACOMBER Bennie D., August 9th, 1994; USPTO 5,352,295, YI Chou, October 4th 1994; USPTO 5,235,945, TESTEA Goerge, August 17th, 1993, EP 1016 785 A1, WO 98/53210, Song Jungyan, May 15th 1998). Actually, almost all rotary vane type engines produce very high torque because the combusted gas expands right against the hot section vane (37,63,100,119), which is the arm length of the generated torque. Therefore, not only is the crankshaft unnecessary, but when comparing engines of equal volumes, the power leverage on the drive shaft of a rotary engine is greater than that of a corresponding reciprocating engine. However, here too, there is room for improvement; the present invention overcomes the drawbacks and limitations of todays power and refrigeration cycles by proposing and implementing new high efficiency thermodynamic cycles (151abceh, 151abcdfh, 151abcgh)

Rotary Engine with Improved Thermodynamic Cycle

The present invention combines the advantages of Otto and Diesel cycles at intake, compression and combustion phases of the thermodynamic cycle by limiting the peak combustion temperature. The present invention also claims an expanded power stroke that greatly improves power extraction and efficiency. With a proper thermodynamic and geometrical match of the compressor and turbine working chambers, it is shown that the expansion process can be improved and lower exhaust pressure and temperature levels can be achieved. A search of the prior art

AMENDED SHEET
23.12.2004

AMENDED SHEET

did not disclose any rotary vane engine patent with separate compression and expansion chambers that considers and provides an expansion process that would take the combusted products further down to ambient pressure levels. The overlook of such thermodynamic cycle improvement is a major source of wasted energy that ultimately translates in engine fuel inefficiency. Accordingly, the present invention provides proper sizing of the compression and expansion chambers, the rotors, and the vanes so as to achieve optimum compression (151ab), combustion (151bce, 151bcdf, 151bcg) and expansion (151eh, 151fh, 151gh). Prior art on compound rotary engine (WO 83/01276, Clarke, John. M., Goloff, Alexander, October 2, 1981) shows a thermodynamic cycle incorporating constant pressure heat addition followed by constant volume heat addition. The difference with the thermodynamic cycle of the present invention lies in the fact that the order of processes, namely the constant pressure heat addition and the constant volume heat addition phases have been switched around. The main reason behind switching the order of the two heat addition processes is as follows. The constant pressure heat addition process in this prior art is actually a low density heat recovery in a recuperator and not a true combustion process per se. As a result, it has to occur at relatively low pressure and temperature values prior to a constant volume heat addition process. When compared with the thermodynamic cycle of the present invention, it can be seen that the aforementioned art loses a substantial amount of net power by switching the order of occurrence of constant volume heat addition and constant pressure heat addition processes. Two prior arts (USPO 5,566,650, Douglas C. Kruse, October 22nd 1996 and GB 2050509A, Haakon Henrik, January 7th 1981) also claim cycles which are at a first sight very close to the thermodynamic cycle of the present invention. However both arts are concerned with piston type geometries which inherently bring out major differences with the rotary vane engine operating method claimed in the present invention. The first main difference concerns the timing of the processes. In a piston type configuration, the compression, combustion and power expansions phases of the thermodynamic cycle have to be completed within a limited time frame to be accomplished within a 180° crank angle rotation. In the present invention, as illustrated in figure 5, the corresponding time frames for the compression (199), combustion (201) and expansion (200) phases have almost

doubled. The extension of thermodynamic processes time frames up to 360° rotor rotation angle, inherently bring many advantages which are not provided by prior arts. There are ample mixing and combustion time (up to 180° compared to 90° in prior arts). As shown in figure 6, the crescent shape geometry at the turbine rotor (204) acts as a nozzle which allows the combustion pressure to be converted to kinetic energy. The combusted flow with sufficient kinetic energy is than able to reach the vane front and come to stagnation without major losses (205). This flow aerodynamic and kinematic supported by the geometry of the rotary vane engines ressembles the initial phases of the turbine inlet nozzles in gas turbine engines. The conversion from pressure to kinetic energy is of upmost importance if expansion up to ambient pressures are sought. The piston like configurations cannot bring such support. Another major difference is the ability of the method of operation cycle of the present invention to provide overlapping power strokes. As each expansion stroke is extended up to 360° rotation angle of the rotor, two power stokes (202 and 203), distanced by 180° phase angle overlap. This also help the flow to expand right down to ambient pressure. This characteristics doesnt exist for aforementioned prior arts that are trying to achieve expansion directly from a combustion pressure built up and within half the time provided by crank angle rotation. Because of their inability to overlap power strokes within two cylinders, piston like configurations also have to devise a complicated mechanism bringing many pistons alltogether, forming a heavy, complicated and loss-prone mechanical device. Another difference between the piston and rotary vane engines is in the way compression (151a, 151b, Figure 5 and 7) is achieved. In other words the compression slope in the P-θ diagram of figure 5 is different for piston type and rotary vane type configurations. In the rotary vane compressors (72,106), the compression is done in a crescent shape cavity which confines the high temperature, high shaft power requiring phase of the compression to a much smaller rotation angle interval. The result is a much efficient nearly isentropic compression process. Figure 7 describe three different thermodynamic cycles. Cycle 1 has a constant volume heat addition followed by constant pressure heat addition followed by maximum power expansion up to ambient pressure: (figure 7, 151abceh). The corresponding air standart efficiency is given as:

$$\eta = 1 - \frac{\gamma}{r_1^{\gamma-1}}\left[\frac{r_c\, r_p^{\frac{1}{\gamma}} - 1}{(r_p - 1) + \gamma r_p (r_c - 1)}\right]$$ where $r_1 = \frac{V_b}{V_a}$ is the compression ratio, $r_c = \frac{V_e}{V_c}$ is the constant pressure heat addition volume ratio also defined as cutoff ratio and $r_p = \frac{P_c}{P_b}$ is constant volume heat addition pressure ratio. γ is gas constant.

Cycle 2 has a constant volume heat addition followed by constant pressure heat addition, followed by constant temperature combustion which is finally followed by maximum power expansion up to ambient pressure: (figure 7, 151abcdfh). The corresponding air standart efficiency is given as:

$$\eta = 1 - \frac{1}{r_1^{\gamma-1}}\left[\left(\frac{r_c\, r_p^{\frac{1}{\gamma}}\, r_T^{\frac{\gamma-1}{\gamma}} - 1}{\frac{1}{\gamma}(r_p - 1) + r_p (r_c - 1)}\right) + \left(\frac{C \ln r_T}{T_a C_v \left[(r_p - 1) + \gamma\, r_p (r_c - 1)\right]}\right)\right]$$, where $r_1 = \frac{V_b}{V_a}$ is the compression ratio, $r_c = \frac{V_d}{V_c}$ is the constant pressure heat addition volume ratio also defined as cutoff ratio and $r_p = \frac{P_c}{P_b}$ is constant volume heat addition pressure ratio. $r_T = \frac{V_f}{V_d}$ constant temperature volume ratio, $C = P_d V_d = P_f V_f$ constant temperature property, C_v, γ are gas constants.

Cycle 3 has a constant volume heat addition followed by constant temperature combustion, followed by maximum power expansion up to ambient pressure: (figure 7, 151abcgh). The corresponding air standart efficiency is given as:

$$\eta = 1 - \frac{1}{r_1^{\gamma-1}}\left[\left(\frac{r_p^{\frac{1}{\gamma}}\, r_T^{\frac{\gamma-1}{\gamma}} - 1}{\frac{1}{\gamma}(r_p - 1)}\right) + \left(\frac{C \ln r_T}{T_a C_v (r_p - 1)}\right)\right]$$, where $r_1 = \frac{V_b}{V_a}$ is the compression ratio, $r_p = \frac{P_c}{P_b}$ is constant volume heat addition pressure ratio. $r_T = \frac{V_g}{V_c}$ constant temperature volume ratio, $C = P_c V_c = P_g V_g$ constant temperature property, C_v, γ are gas constants.

OBJECT OF THE INVENTION

One of the objects of this invention, is to increase the thermal efficiency above levels reached by today's heat engines. This is achieved by implementing a longer power extraction phase (151eh, 151fh, 151gh) and by realising high compression ratios with less shaft power input, by processing the fluid through a smooth crescent shape constriction (72 and 49 and 53). The operational and maintenance costs are also minimised, as maximum peak temperature is limited. All together, the present invention discloses an efficient, powerful, compact, simple and reliable heat engine.

For the compound engine configuration of the instant invention, rotary components and gas turbine engine components have been matched with each other. The objective is to combine the high efficiency and "no-stall" characteristics of internal combustion engines with the high mass flow, smaller size and lighter weight characteristics of the gas turbine engines. Another objective is to eliminate the long, heavy and cumbersome concentric shafts and reduction gears that are present in today's turbofan, turboprops and turbojet engines. By simplifying the mechanical links and by integrating low mass flow rotary components, the implementation of high efficiency reheat and intercooling systems have become extremely feasible..

SUMMARY OF THE INVENTION

The solo configuration (Figures 1,2,3,4) of the invention relates to a rotary vane type machine comprising a compressor (10,19;46,48) and a turbine (36,43;57,59) housing, each having a crescent shape cavity. Each of these housing is receiving an eccentrically placed rotor (4,11,89,96,130,117) equipped by a radially movable single sliding vane (50,63) arranged in the rotor. Within each housing, depending on the rotational position of the sliding vanes, forms a plurality of working chambers (49, 53, 60, 66, 72) each of the said chambers, delimited by the inner peripheral surface of the housing (48, 59), the outer peripheral surface of the rotor (90, 98) and the side surface of the vane (16, 37). With such configuration, the solo use of the turbo-rotary engine of the invention overcomes the limitations of conventional internal combustion engines and enables significant improvement in power, torque and efficiency. For a power range of 500 kW-5MW, the present invention (Figures 8,9,10) extends the

efficient but narrow operating range of the gas turbine engine by mechanically decoupling and eliminating the long shaft drive between the expander (turbine) and the turbo-compressor. Each said fan (153,155) and compressor group (158,161,182,197) is allowed to be driven at its own speed, by its own rotary turbine (154,156,157,162, 181, 196) wherein, amounts of combustion fuel and air is delivered is dictated by the instantaneous compressor load requirements. Turbines (170, 171, 178, 191) drives rotary compressors (164,166,168,179,195,190) that pumps high pressure fluid to respective rotary turbines. Therefore the present invention overcomes some of the off-design limitations of conventional gas turbine engines. Because rotary vane compressors and turbines are partial admission components, they have low mass flow rate requirements and it also becomes extremely cheap and useful to equip the system with intercoolers (193) and reheat (198) systems. Other features, advantages, and applications of the invention will be apparent from the following descriptions, the accompanying figures, and from the claims.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention is illustrated hereinafter through preferred and alternative embodiments wherein:

Figure 1, is a schematic isometric view of a rotary engine where compressor (10) and turbine (43) housings are arranged in tandem. The gas transfer from the compressor to the turbine is sequenced by a rotor synchronised (24, 25, 26, 27, 8, 7, 6) cyclo-valve. Combustion occurs within the turbine expansion chamber. The shaft (29) is directly linked to the turbine rotor (4). The cyclo-valve (112, 113) is housed (5) between the turbine (43) and compressor (10) casings.

Figure 2, is a schematic top view of the rotary engine of Figure 1, sectioned at mid height of the turbine housing. Isometric view of the cyclo-valve (69, 70, 71) is added. The bolt holes (47, 77) in the outer housing (78) is used to seal upper (111) and lower plates. The inner housing (79) is tight fit within the outer housing (78).

Figure 3, is a schematic isometric view of a rotary engine where compressor and turbine housings are arranged in tandem. The gas is transferred from the compressor to the turbine through an intermediary cyclo-combustion chamber (74) synchronised with the rotational speed of the compressor and the turbine. Expansion occurs within the turbine chamber (103).

Figure 4, is a schematic isometric view of a rotary engine where compressor (122) and turbine (121) housings are arranged back-to-back thereof, the compressor rotor (126) is coaxial with the turbine rotor (117). Combustion occurs externally within a cyclo chamber (114) and expansion occurs within the turbine.

Figure 5, P-θ diagram and angular range of compression, heat addition and expansion processes.

Figure 6, Force coupling and nozzle like expansion at the inlet of the rotary vane turbine.

Figure 7, high efficiency, high power, low peak temperature new thermodynamic cycles (151abceh, 151abcdfh, 151abcgh) pertaining to the invention.

Figure 8, turbo-rotary-fan compound engine with high inlet (152) mass flow rate

Figure 9, turbo-rotary-prop compound engine

Figure 10, turbo-rotary compound engine for helicopter or power applications.

Figure 11, Compound use of rotary and Brayton thermodynamic cycles

DETAILED DESCRIPTION OF THE INVENTION

The embodiments of the invention for which an exclusive privilege or property is claimed are described as follows:

Figure 1 and Figure 2 depict a preferred embodiment of the internal combustion rotary vane engine where combustion occurs within the turbine chamber (66). Figure 3 depicts a different preferred embodiment of an external combustion (74, 92) rotary vane engine where combustion starts in a chamber (72) prior to entry within the

turbine expansion chamber (103). The rotary combustion engine casing (14, 21, 35, 44) comprises at least one rotary compressor unit wherein, said unit has inner (19, 48) and outer (46) housings. The engine casing also comprises at least one rotary turbine unit wherein, said unit has inner (59) and outer (57) housings. Said housings are surrounded by a liquid cooled jacket (18, 41, 45). For example, water may be used as coolant. The housings have each, a circularly cylindrical (3, 13) rotor (12,128), rotatably and eccentrically mounted. The said rotary engine breathes through the intake (20) and exhaust (34,116) ports. The compressor rotor outer boundary (83,126) and turbine rotor outer boundary (98) are sealingly (86,101) mounted (90, 95) tangent to the chamber inner peripherals (88, 97). Accordingly, the respective rotor outer boundary and the chamber inner peripheral are osculating at their common tangency plane (90, 95).

The sliding vane also carries many slots (14, 38, 40, 52, 64, 84, 124, 120) that accommodates seals. The radially outer vane tips (51,86,101,109) are always in a natural contact with the housing inner peripheral (88, 97).

Intake chamber (53) is receiving the fluid from intake port (20), said fluid is either air, or any other working gas, or any other liquid-gas mixture. Said fluid, is compressed by the compressor rotor (12, 89) and the single rigid sliding vane (50, 87) which is sealingly (86) and movably mounted within the rotor groove. The sliding vane is contoured (127) to fit the said groove. By placing the sliding vane (16, 50, 87) within the compressor housing, a plurality of working chambers (72, 49, 53 and 106, 107, 108) are sequentially created within the crescent shaped cavity, delimited by the compressor housing inner peripheral (88) and the rotor outer surface (83). When compared with gas turbine engine compressor, the rotary compression work is at least 2-15% more efficient as the fluid is sealed within the closed control volume (49, 107). The compression work within the crescent shape is smooth and gradual and therefore the compression is nearly isentropic. As a result of the rotation of the rotor, a periodic sequence of compressed fluid is delivered to the exhaust port (56, 76) of the compressor housing. The rotary combustion engine compressor casing is sealed at its opposite ends by bolted (9) plates (10). One of the compressor end plates is apertured at its centerline to allow for the drive shaft.

AMENDED SHEET
23.12.2004

Figures 1, 2 and 3 depict preferred embodiments where compressor and turbine housings are arranged in tandem. For such configurations, power and torque transfer and synchronization of compressor and turbine rotors (12, 3) and valve rotation (67, 69) are achieved with drive shafts (31, 8, 6, 69), gears (33, 28, 26, 24) or other transmission mechanisms (7, 25, 30), and bearings (not shown) supported by the engine casing (32, 21, 22, 23, 27). For the preferred embodiment depicted in Figure 4, power transfer between the turbine and the compressor is much simpler, as the compressor (129) and turbine (117) rotors are directly coupled to the drive shaft (110). The shaft is journalled in bearings supported by the engine casing (121,122). The back-to-back compressor and turbine configuration is compact and lightweight as no gear, pulley and auxiliary power and torque transfer equipment is used. A fluid transfer passage (114) connects the exhaust port (76) of the compressor and the intake port (73) of the turbine. In the external combustion engine depicted in Figure 3, the said transfer passage includes a combustion chamber (74, 92) which is periodically pressurised by gas flow from the compressor exhaust port. Downstream of said compressor exhaust port there is at least one check valve and/or at least one cyclo-valve (67, 68, 93) , operating between open and closed positions, in timed sequence with the passage of the turbine sliding vane (100). The closed position (94) of the said valve prevents gas flow from the combustion chamber into the compressor. The cyclo-valve comprises a tubing (70) having inlet (115) and exhaust ports (71) and a rotatably and sealingly mounted slotted cylinder (69) within the said tubing.

Two firing cycles occur per rotor revolution. As one firing takes place (91), new cycles (92, 107, 108) are preceding the present firing and at least one old cycle (60, 99) is terminating thereof, a smooth operation is assured. The rotary turbine unit is similar to the compressor unit but its size differs. Working chambers (60, 66, 103, 99), belonging to the turbine are delimited by the housing inner peripheral (97), the rotor outer surface (98) and the side surface of the sliding vane (37). For Figure 3, the combustion working chamber (74) and the expansion chamber (103) are separate but linked. Thus, the turbine housing is allowed to run at a reduced temperature. A periodic sequence of expanded fluid is delivered from the exhaust port (34) with each rotation of the turbine rotor in response to high pressure and temperature gas

expansion in the said turbine. The exhaust gas pressure is lowered to about local ambient pressure values to allow maximum shaft work extraction and increase in thermal efficiency. As shown in Figure 7, the maximum volume of the turbine expansion chamber (66) is sized such that the combusted gas pressure (151e, 151f, 151g) is expanded to local ambient pressure (151h). The height of the turbine inner housing (36) and the turbine rotor (117) is sized in such a way that, the pressure of the gas, as it is being transferred from the compressor chamber (72) is maintained to about a constant high value (151b). The turbine casing (35) is sealed at its opposite ends by bolted (42) plates (43). One of the said plates is apertured at its centerline, to accommodate the drive shaft (31) protruding therefrom.

A fuel or a fuel/atomiser mixture (75) is supplied to said combustion chamber (74, 66) using commercially available fuel injection or fuel aspiration means. The fuel/oxidizer mixture is ignited using commercially available spark (91) or pressure ignition means. For each 360 degrees of rotation of said compressor rotor, there are two complete and consecutive cycles of intake (108, 151a), compression (92, 151ab), combustion (74, 151bce; 151bcdf; 151bcg), power (66,103, 151eh; 151fh; 151gh) and exhaust (60, 99, 151h) phases.

The thermodynamic cycle associated with the intake, compression, combustion, expansion and exhaust phases of the rotary engine contains innovations when compared to the Otto, Brayton, Diesel or more recent increased expansion cycles proposed in prior arts (PCT WO 02/090738, DUNCAN, Ronnie J., November 14th, 2002; USPTO 5,341,771, RILEY, Michael B., August 30th, 1994). At intake, compression and combustion phases, the present invention combines the advantages of Otto and Diesel thermodynamic cycles. It is well known that for a given compression ratio, the ideal Otto cycle currently provides the most efficient combustion/expansion process since it combines high peak temperature during the isochoric (constant volume) heat addition, while still keeping an acceptable mean chamber temperature. However, high peak combustion temperatures can cause auto-ignition of a portion of fuel-air mixture, resulting in engine knocks. Diesel is an improvement of the Otto cycle as it provides higher useful compression ratios and isobaric (constant pressure) heat addition and do not have knock problem as air

AMENDED SHEET
23.12.2004

AMENDED SHEET

alone is present during the compression process. The high compression ratio makes Diesel engines more fuel-efficient but for this same reason, they also become much heavier. Compared to the Otto cycle. Diesel cycle also delivers less power for the same displacement. For the compression and combustion phases of the cycle, the ideal would be to follow a limited combustion pressure cycle that would first use a combined isochoric heat addition followed by isobaric and/or isothermal heat additions. As mentioned in a prior art, (USPTO, 5,566,650, KRUSE, Douglas C., October 22nd, 1996) such hybrid engine process has been developed (Texaco TCCS, Ford PROCO, Ricardo, MAN-FM and KHD-AD) but they have been proven impractical. The rotary engine of the present invention naturally follows the above-described limited peak cycle (151bce; 151bcdf; 151bcg) multi-step (isochoric, isobaric and/or isothermal) combustion phases.

By limiting the peak combustion pressures, the present invention also provides an expanded power stroke that improves power extraction (151eh; 151fh; 151gh). A search of the prior art did not disclose any patents that considers a thermodynamic heat engine cycle, whether it be reciprocating or rotary, that jointly proposes a limited peak combustion pressure and an expansion phase where the pressure exhausts (180, 192) to about ambient pressure.

One of the drawbacks of the current gas turbine engines are their highly sensitive stall characteristics always placed close to the high performance region. Furthermore, shaft and aero-thermodynamic coupling and feedback loop between the compressor and the turbine, only allows a narrow, high efficiency operational band. The present invention provides a practical and effective means of adding higher degrees of freedom to the gas turbine engines by eliminating the shaft coupling between fans, compressors and turbines. The compound engine (Figures 8, 9, 10) of the present invention combines the efficiency of the heat engines with the compactness, lightweight and high power characteristics of gas turbine engines. Heat engines can produce kilowatts of power at high power densities and efficiencies but they are heavy and of relatively large sizes. After 50MW, most of thermodynamic, scaling and cost considerations have favoured large size gas turbine engines. 100 MW gas turbine combined cycle power plant costs about 500 USD per kW, whereas

10 MW power plant costs 750 USD per kW and 1 MW power plant costs 1000 USD per kW. For small power range (0.1-10 MW), internal combustion engine becomes highly competitive despite their size. weight and high maintenance cost. The present invention overcomes the limitations of both small gas turbine engines and large internal combustion engines and meets the modularity, high efficiency, mobility weight and cost requirements of today's modern power and propulsion applications.

A number of prior arts combine entire rotary combustion engines with gas turbine engine components (Davorin Kapich, USPTO 5,471,834, December 5th 1995, Whurr John, USPTO 5,692,372, December 2nd 1997). There is also a prior art which combines gas turbine engine components with positive displacement rotary components (Clarke, John M., Goloff, Alexander, WO83/01276, April, 14th 1983). However all of them use and combines conventional components readily available without attempting to further improve the overall thermal efficiency of positive displacement components as is the case with the present invention. Accordingly, WO83/01276 rely on a multistage expansion means where the task to lower combusted gas pressure up to ambient is still given to the conventional turbine (Brayton cycle). Whereas, as it can be seen from Figure 11, in the present invention ,expansion to ambient pressures are both assured by the novel rotary vane thermodynamic cycle and the Brayton cycles. Also in WO83/01276 the peak temperature belongs to a constant volume combustion phase which is then followed by maximum expansion phase which clearly shows that this art is based on the well known Atkinson cycle (Page 7 line 30, WO83/01276) with further addition of recuperation, whereas the present invention rely on a novel cycle which combines temperature limited constant volume combustion followed by constant pressure heat addition phases followed by a maximum expansion phase. The thermodynamic cycles are therefore fundamentally different in the peak temperature limiting processes. The two prior arts (USPTO 5,471,834 and USPTO 5,692,372) also do not emphasize on improving the efficiency of rotary engines. None of these prior arts is based on the coupling (figure 11) of two highly efficient thermodynamic cycles as is proposed in the present invention.

The invention provides a preferred embodiment coupling the high mass flow gas turbine engine components with partial admission rotary vane components which

have relatively low mass flow rates. This is achieved by a mechanical coupling (162, 165, 167) of the rotary engine components with conventional gas turbine engine components. The turbo-rotary compound engines (Figures 8. 9. 10) of the invention. eliminates conventional long and heavy concentric shafts and disclose a novel configuration where conventional rotational wing (186), fans (153, 155) compressors (158, 161, 182, 197) and turbines (170, 171, 178, 191) are only aero-thermodynamically coupled with each other. In this invention, rotary turbines (154, 156, 157, 163, 181, 188, 196) drive said rotational wing, fans and compressors. Conversely, said conventional turbines drive single or a plurality of rotary compressors (163,164, 166, 168, 179, 195, 190). Rotary compressors supply compressed fluid via flexible high-pressure connections (172, 187, 189, 194). The new thermodynamic cycle of figure 7 is used in conjunction with the Brayton cycle as illustrated in figure 11. Power generated by rotary vane turbines drive the axial compressors (159, 160) and fan units; $\dot{W}_{2\to 3\,Axial\,Compressor} = \dot{W}_{4r\to 5r\,Rotary\,Turbine}$ and vice versa, part of the power generated by axial turbines drives the rotary vane compressor; $\dot{W}_{0\to 2r\,Rotary\,Compressor} = \dot{W}_{4\to 5\,Axial\,Turbine}$. Relatively low mass flow will move through such connection as both rotary compressors and turbines are partial admission machines. Low mass flow also favours the efficient use of intercooler (193) and reheat (198) systems, giving a further boost to thermal efficiency. The compound engine of the invention combines the thermal efficiency of the rotary internal engine cycle and the high mass flow, compact size and lightweight of the gas turbine engines. Such a compound cycle propulsion engine may comprise propellers, conventional fans, contra-rotating fans, hub-turbine driven fans (177, 176), contra-rotating hub-turbine driven fans (175, 174, 184,185), axial and/or centrifugal compressor stages, combustion chambers (169, 173), axial and/or centrifugal turbine stages, rotary compressors and turbines, intercoolers and reheaters. The current design is versatile and simple, therefore well adapted to turboprop, turbofan, marine and land based power production and refrigeration applications. The subject design can also be applied to geothermal power plants.

As is demonstrated by the breath of this description, the range of application for the compound and solo use of the turbo-rotary engine provided by the invention is truly

vast. In particular, the scope of the present invention includes hybrid turbo-rotary engines where conventional axial and/or radial turbines drive both conventional axial and/or centrifugal compressors and rotary compressors. Also included in the present invention, hybrid applications where conventional axial and/or centrifugal compressors are driven by both conventional axial and/or radial turbines and rotary turbines. While the description cannot address each and every application, it is intended to indicate the extensive capabilities contemplated by the invention.

EPO - DG 1

24 12. 2004

68

CLAIMS

The embodiments of the invention for which an exclusive privilege or property is claimed, are defined as follows:

1. A method of operating, controlling, and monitoring a sliding vane rotary heat engine for providing limited temperature constant volume combustion followed by constant pressure combustion which is then followed by a power expansion phase up to ambient pressure. Said method comprising the following sequential steps of a thermodynamic cycle pertaining to a sliding vane rotary heat engine:

An intake phase providing fluid to a positive displacement sliding vane rotary vane compressor at ambient pressure.

A fluid compression process (151ab) within a sliding vane rotary compressor characterized by a long time interval extending up to 360 degrees of rotation of the compressor rotor (199). Said compression process is further improved as it is achieved within a crescent shape cavity (72, 106) whose volume gradually decreases as the single sliding vane squeezes out the fluid to an external combustion chamber (201) though the synchronized timing of on/off valves.

A limited temperature constant volume combustion process (151bc) receiving compressed fluid from the sliding vane rotary compressor and using a predetermined fuel/oxidiser mixture ignited within a combustion chamber (94) external to rotary vane compressor and rotary vane turbine. Wherein said limited temperature constant volume combustion process is characterized by a long mixing and combustion time (201) up to 180 degrees of the rotation of the compressor rotor. Said external combustion chamber delivering combusted products to the sliding vane rotary turbine (expander) though the synchronized timing of on/off valves.

The fluid is subsequently subjected to a constant pressure combustion process (151ce) followed by power extraction phase (151eh) within a sliding vane rotary turbine (expander). The power expansion phase is also characterized by the presence of two overlapping power strokes (202, 203) allowing the expansion to continue for a long time interval extending up to 360 degrees of the turbine rotor

AMENDED SHEET

(200). Expansion phase also characterized by the burned gas pressure reaching ambient pressure when the turbine expansion chamber volume reaches its maximum.

An exhaust phase ejecting combusted fluid from the sliding vane rotary turbine (expander).

2. A method of operating, controlling, and monitoring a sliding vane rotary heat engine for providing limited temperature constant volume combustion followed by constant pressure combustion which is then followed by a constant temperature combustion process. Subsequently, a power expansion phase starts and continues up to ambient pressure. Said method comprising the following sequential steps of a thermodynamic cycle pertaining to a sliding vane rotary heat engine:

An intake phase providing fluid to a positive displacement sliding vane rotary vane compressor at ambient pressure.

A fluid compression process (151ab) within a sliding vane rotary compressor characterized by a long time interval extending up to 360 degrees of rotation of the compressor rotor (199). Said compression process is further improved as it is achieved within a crescent shape cavity (72, 106) whose volume gradually decreases as the single sliding vane squeezes out the fluid to an external combustion chamber (201) though the synchronized timing of on/off valves.

A limited temperature constant volume combustion process (151bc) receiving compressed fluid from the sliding vane rotary compressor and using a predetermined fuel/oxidiser mixture ignited within a combustion chamber (94) external to rotary compressor and rotary turbine. Wherein said limited temperature constant volume combustion process is characterized by a long mixing and combustion time (201) up to 180 degrees of the rotation of the compressor rotor. Said external combustion chamber delivering combusted products to the sliding vane rotary turbine (expander) though the synchronized timing of on/off valves.

The fluid is subsequently subjected to a constant pressure combustion process (151cd) followed by a constant temperature combustion process (151df) which then

followed by a power extraction phase (151fh), all occurring within a sliding vane rotary turbine (expander). The power expansion phase is also characterized by the presence of two overlapping power strokes (202, 203)allowing the expansion to continue for a long time interval extending up to 360 degrees of the turbine rotor (200). Expansion phase also characterized by the burned gas pressure reaching ambient pressure when the turbine expansion chamber volume reaches its maximum.

An exhaust phase ejecting combusted fluid from the sliding vane rotary turbine (expander).

3. A method of operating, controlling, and monitoring a sliding vane rotary heat engine for providing limited temperature constant volume combustion followed by a constant temperature combustion process. Subsequently, a power expansion phase starts and continues up to ambient pressure. Said method comprising the following sequential steps of a thermodynamic cycle pertaining to a sliding vane rotary heat engine:

An intake phase providing fluid to a positive displacement sliding vane rotary vane compressor at ambient pressure.

A fluid compression process (151ab) within a sliding vane rotary compressor characterized by a long time interval extending up to 360 degrees of rotation of the compressor rotor (199). Said compression process is further improved as it is achieved within a crescent shape cavity (72,106) whose volume gradually decreases as the single sliding vane squeezes out the fluid to an external combustion chamber though the synchronized timing of on/off valves.

A limited temperature constant volume combustion process (151bc) receiving compressed fluid from the sliding vane rotary compressor and using a predetermined fuel/oxidiser mixture ignited within a combustion chamber (94) external to rotary vane compressor and rotary vane turbine. Wherein said limited temperature constant volume combustion process is characterized by a long mixing and combustion time (201)up to 180 degrees of the rotation of the compressor rotor. Said external

combustion chamber delivering combusted products to the sliding vane rotary turbine (expander) though the synchronized timing of on/off valves.

The fluid is subsequently subjected to a constant temperature combustion process (151cg) which then followed by a power extraction phase (151gh), all occurring within a sliding vane rotary turbine (expander). The power expansion phase is also characterized by the presence of two overlapping power strokes (202, 203) allowing the expansion to continue for a long time interval extending up to 360 degrees of the turbine rotor (200). Expansion phase also characterized by the burned gas pressure reaching ambient pressure when the turbine expansion chamber volume reaches its maximum.

An exhaust phase ejecting combusted fluid from the sliding vane rotary turbine (expander).

4. A sliding vane rotary heat engine, having an intake (20), a rotary compressor equipped by a single sliding vane (50), a combustor (94) external to said rotary compressor and rotary turbine, a rotary turbine (expander) equipped by a single sliding vane (63) and an exhaust (116). Said sliding vane rotary heat engine is characterised by its method of operating, controlling and monitoring as described by claim 1 or claim 2 or claim 3.

5. A compound propulsion engine comprising:

a). An axial compressor (182) wheel driven by sliding vane rotary turbine (181) (expander). A plurality of compressor blade means attached to said compressor wheel having a shaft with an axis defining an axial direction such that flow exits said compressor blades in essentially said axial direction. Said axial compressor, sliding vane rotary expander and interconnecting shaft defining altogether a primary spool. Axial compressor to receive and further compress ram air (183) , ram air defining stage 1 compressed air and axial compressor exit air defining stage 2 compressed air. Combustor (173) placed downstream of axial compressor is used for heat addition to stage 2 compressed air so as to produce stage 3 combusted efflux.

AMENDED SHEET

b) An axial turbine (178) wheel driving sliding vane rotary compressor (179). A plurality of turbine blade means attached to said turbine wheel having a shaft with an axis defining an axial direction such that flow exits said turbine blades in essentially said axial direction. Said axial turbine, sliding vane rotary compressor and interconnecting shaft defining altogether a secondary spool. Secondary spool characterized as not being connected to primary spool through a driving shaft. Axial turbine positioned to receive and being driven by said stage 3 combusted efflux.

c) Said sliding vane rotary compressor arranged to receive a portion of ram air or a portion of said stage 2 compressed air. Said sliding vane rotary compressor further compressing said portion of compressed air for producing a secondary flow. Said secondary flow directed through fluid transfer passages (172) defined as connecting flow lines between exhaust port of rotary vane compressor and intake port of rotary vane turbine. Said transfer passages comprise open/close valves and heat addition or removal means (193, 198). Secondary flow characterized as fluid flow supplied by sliding vane rotary compressor of said secondary spool to sliding vane rotary turbine of said primary spool.

6. A compound propulsion engine as in claim 5 and further comprising:

a) A second axial turbine (191) defined as axial turbine 2 and placed downstream of said axial turbine defined in claim 5. Said axial turbine 2 driving a second sliding vane rotary compressor (190) defined as rotary compressor 2 through a shaft. Said axial turbine 2, rotary compressor 2 and interconnecting shaft defining altogether secondary spool 2

b) Said rotary compressor 2 arranged to receive a portion of ram air or a portion of stage 2 compressed air (194). Said rotary compressor 2 further compressing said portion of compressed air for producing a secondary flow defined as secondary flow 2.

AMENDED SHEET
23.12.2004

AMENDED SHEET

Said secondary flow 2 directed through fluid transfer passages defined as connecting flow lines (187) between exhaust port of rotary compressor 2 and intake port of a second sliding vane rotary turbine (188) defined as rotary turbine 2. Said transfer passages comprise open/close valves and heat addition or removal means (193, 198). Secondary flow 2 characterized as fluid flow supplied by rotary compressor 2 (190) of said secondary spool 2 to rotary turbine 2 (188).

7. A compound propulsion engine as in claim 5 and further comprising:

a) An axial fan (155) wheel driven by sliding vane rotary turbine (156) defined as rotary turbine 3. A plurality of fan blade means attached to said fan wheel having a shaft with an axis defining an axial direction such that flow exits said fan blades in essentially said axial direction. Said axial fan defining a propulsion fan defined as fan 3. Said fan 3 placed upstream of said axial compressor (158) defined in claim 5. Said fan 3, rotary turbine 3 and interconnecting shaft defining altogether primary spool 3. Said primary spool 3 characterized as not being connected to any other primary or secondary spools through a driving shaft. Intake port of rotary turbine 3 receiving a portion of said secondary flow produced by said rotary vane compressor (168) defined in claim 5.

b) An axial fan (153) wheel driven by sliding vane rotary turbine (154) defined as rotary turbine 4. Said axial fan defining a propulsion fan defined as fan 4 and rotating in the opposite direction of rotation of fan 3. Said fan 4 placed upstream of said fan 3. Said fan 4, rotary turbine 4 and interconnecting shaft defining altogether primary spool 4. Said primary spool 4 characterized as not being connected to any other primary or secondary spools through a driving shaft. Intake port of rotary turbine 4 receiving a portion of said secondary flow produced by said rotary vane compressor (168) defined in claim 5.

AMENDED SHEET

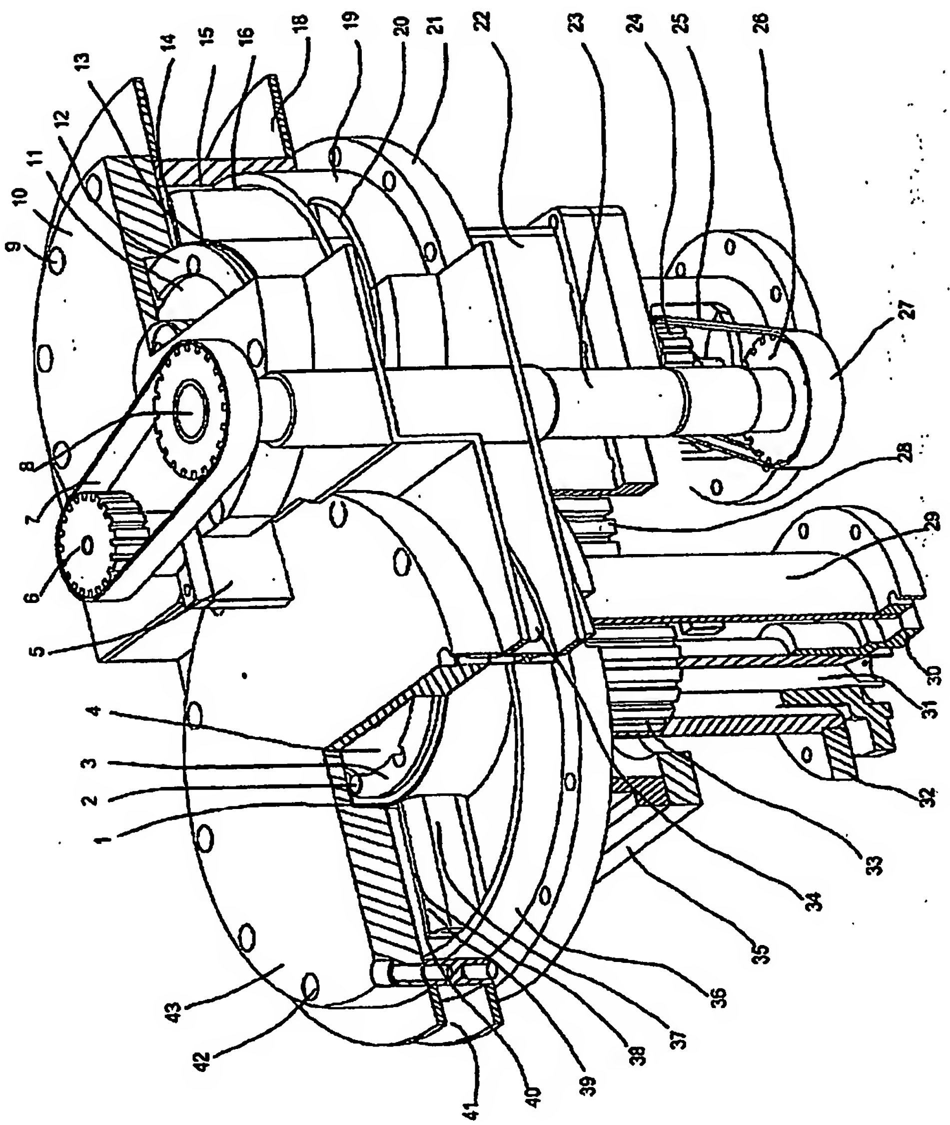
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
34
35
36
37
38
39
40
41
42
43

FIGURE 1

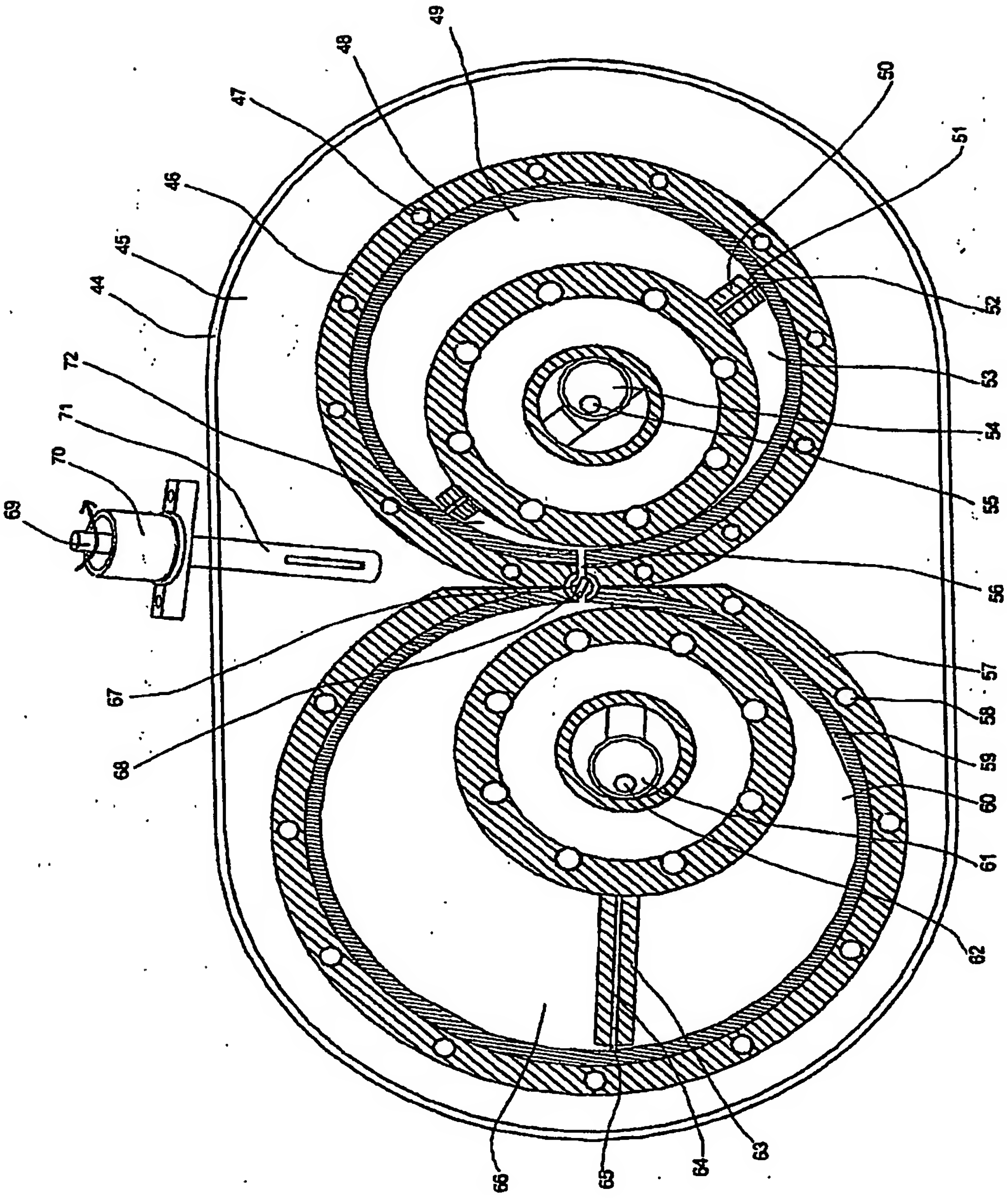
44
45
46
47
48
49
50
51
52
53
54
55
56
57
58
59
60
61
62
63
64
65
66
67
68
69
70
71
72

FIGURE 2

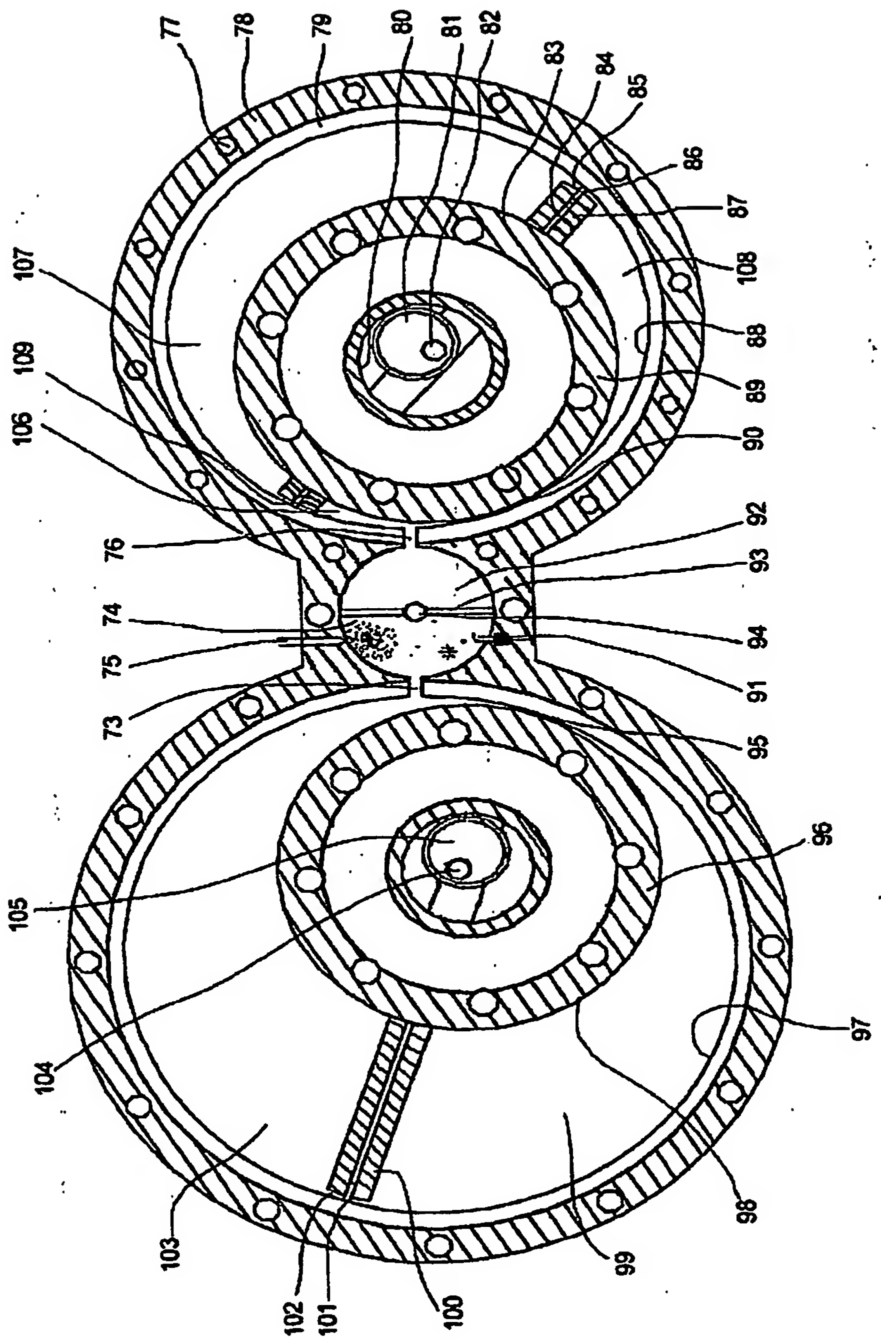
77
78
79
80
81
82
83
84
85
86
87
108
88
89
90
92
93
94
91
95
96
97
98
99
100
101
102
103
104
105
73
75
74
76
106
109
107

FIGURE 3

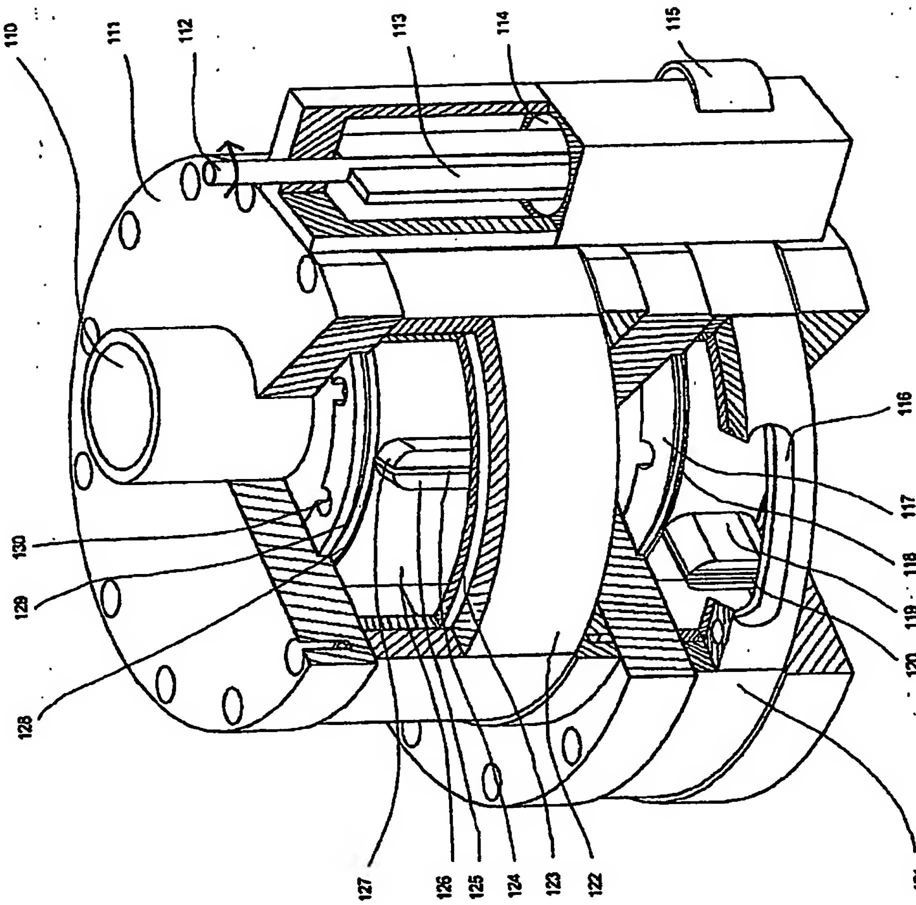
110
111
112
113
114
115
116
117
118
119
120
121
122
123
124
125
126
127
128
129
130

FIGURE 4

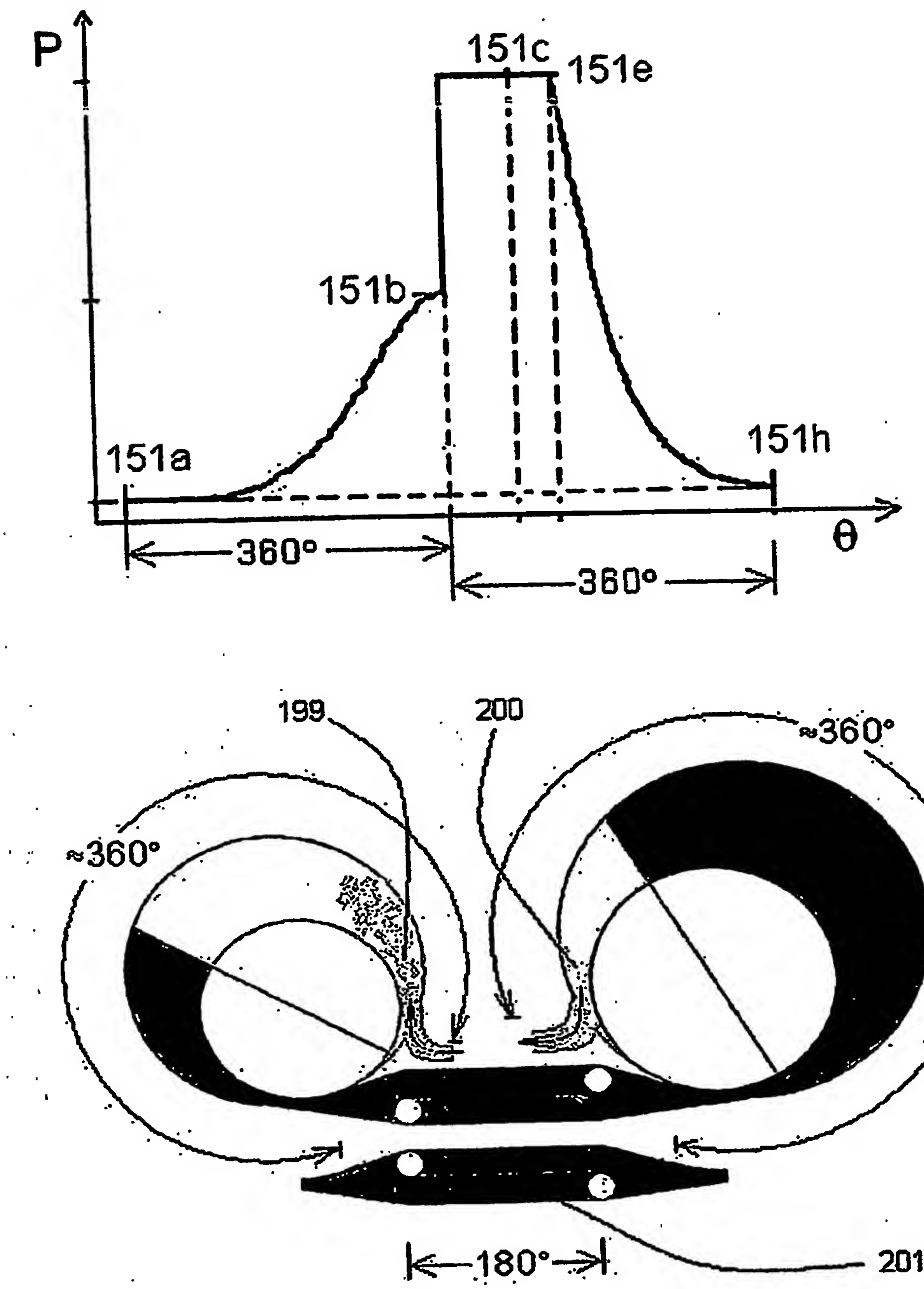

FIGURE 5

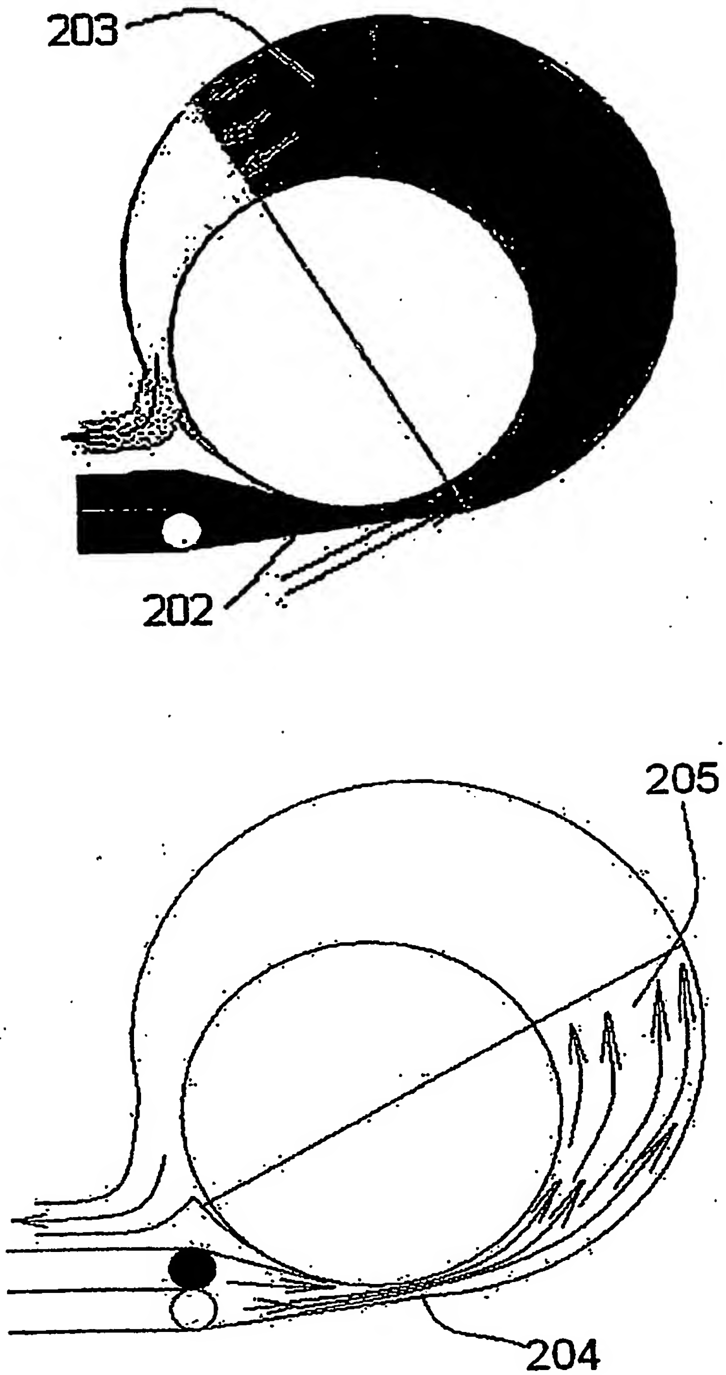

FIGURE 6

REPLACED SHEET
23.12.2004

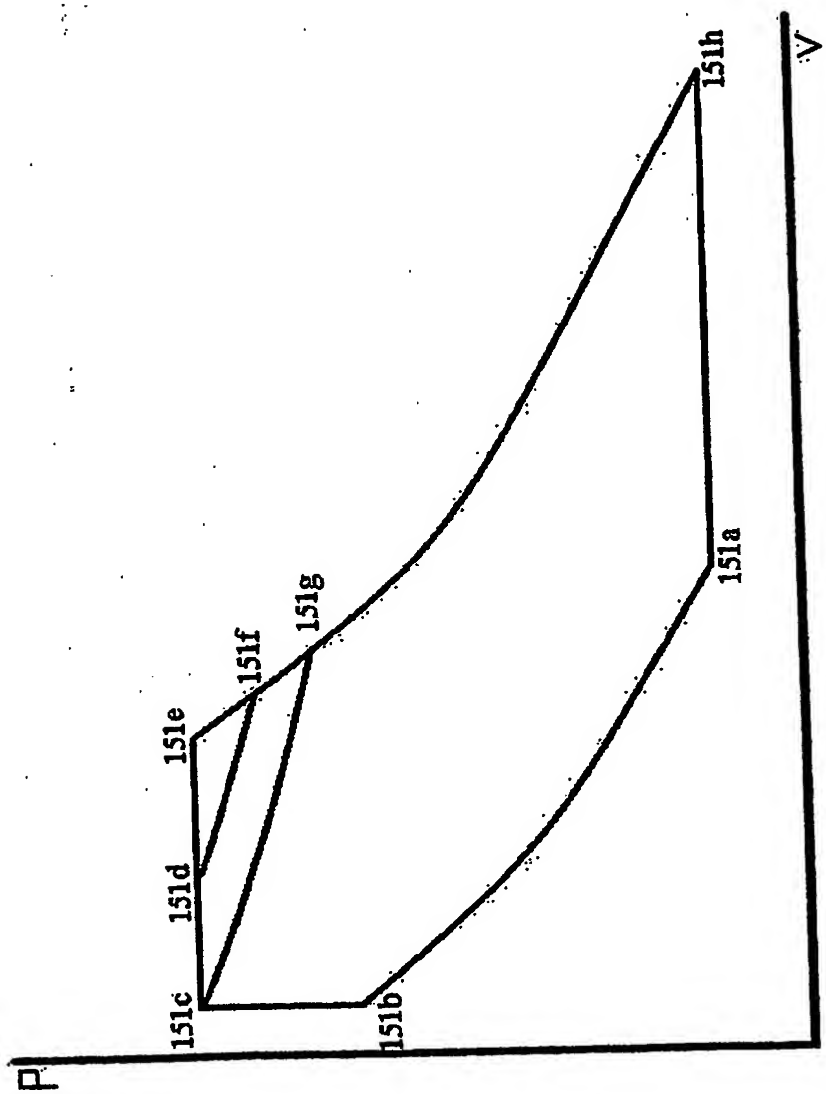

FIGURE 7

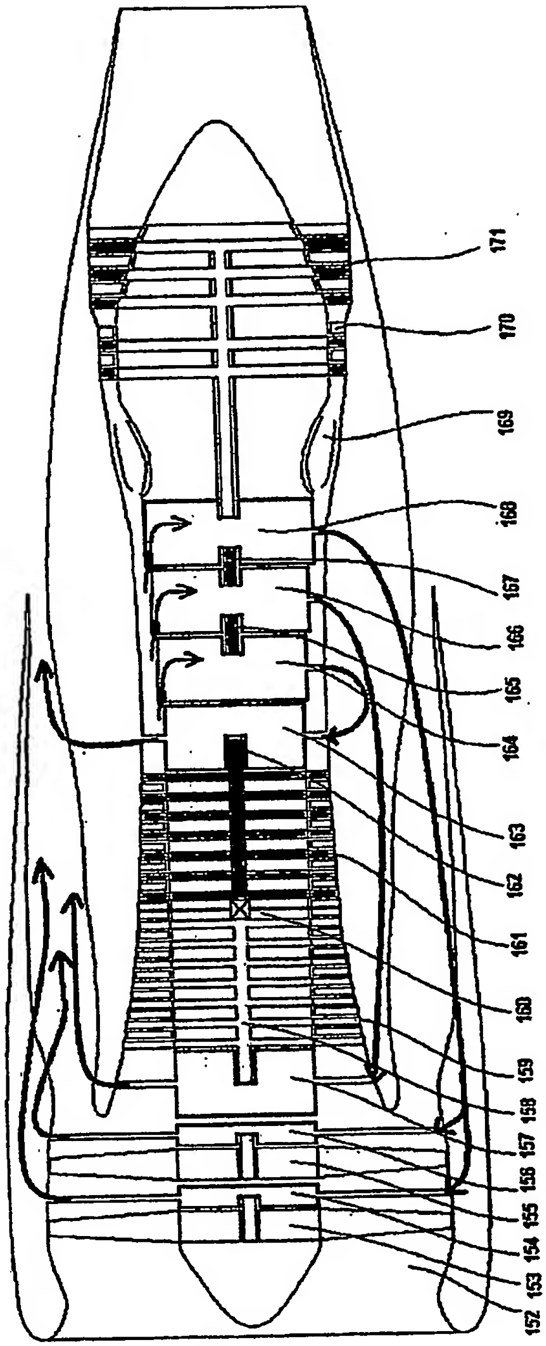
171
170
169
168
167
166
165
164
163
162
161
160
159
158
157
156
155
154
153
152

FIGURE 8

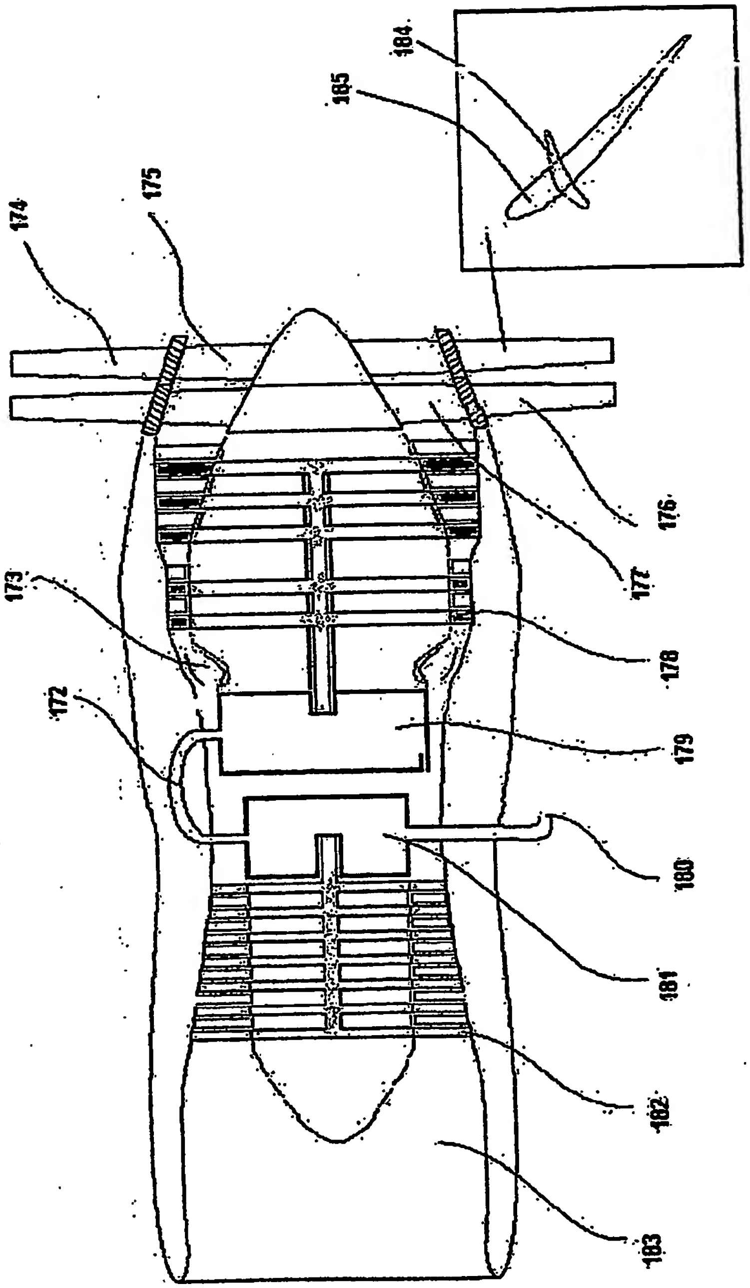

FIGURE 9

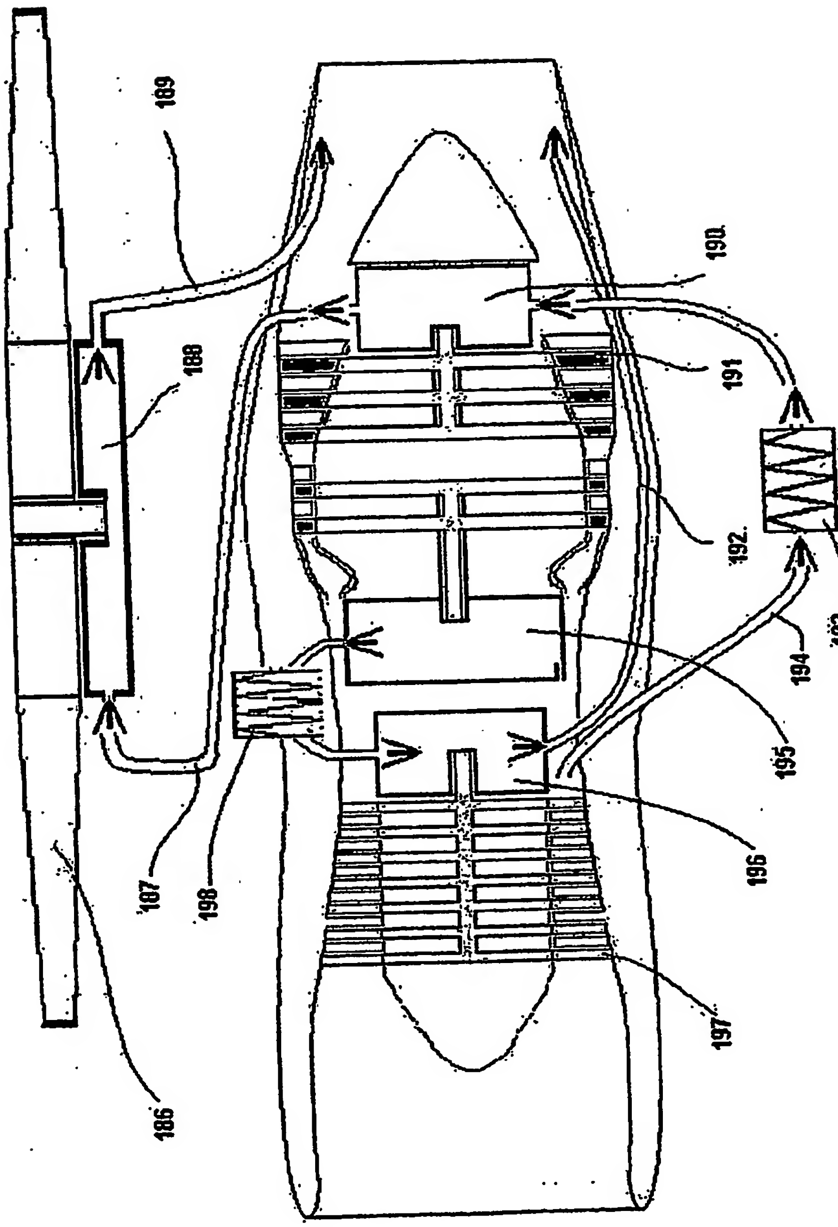

FIGURE 10

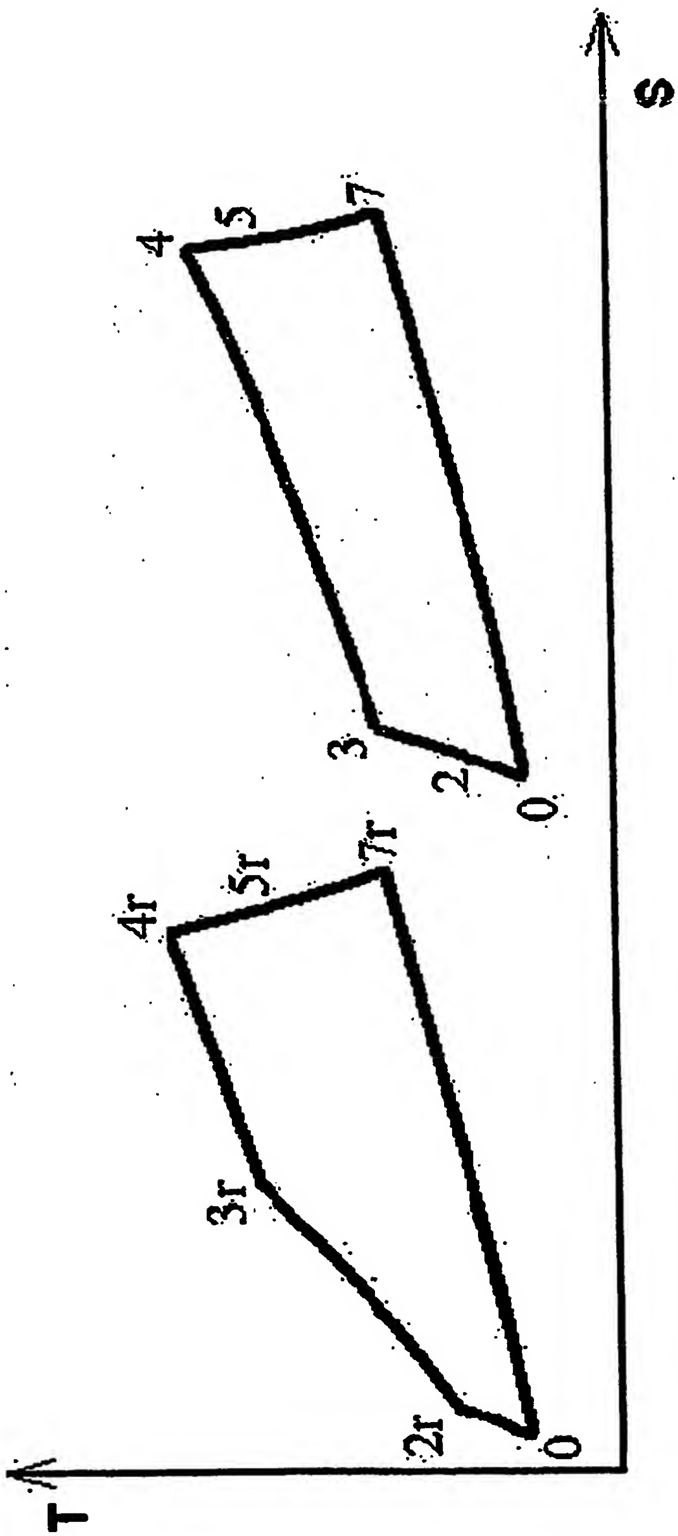

FIGURE 11

ADDED SHEET
23.12.2004

AMENDED SHEET